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Procedia Engineering 31 (2012) 817 – 822
Procedia Engineering
www.elsevier.com/locate/procedia

International Conference on Advances in Computational Modeling and Simulation

Modeling and Simulation of an Air-cooling Condenser under Transient Conditions

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Abstract

Condenser is a major component of the refrigeration and air conditioning system, the performance of the condenser will have a direct influence on the system efficiency. In practice, the condenser always works in varying conditions. So it is essential to investigate its transient response for system design and control strategy. In this paper, a dynamic mathematical model of an air-cooling condenser based on the moving-boundary model has been developed for control strategy, using the zone model. Customary moving-boundary approach applied in heat exchanger simulation always focuses on the refrigerants side but neglect the air side. In order to study the actual behavior of the condenser, both sides of the condenser coil are fully considered in the model for this work. Moreover, to avoid iterative operation, state parameters of the mathematical model are expressed as the functions of the temperature in this paper. Depending on this model, several types of transient conditions such as step changes of the compressor speed, air temperature and air quality flow rate are studied. The simulation results show satisfactory predictions which indicate that this model can be used to formulate a control algorithm.

Keywords: Condenser; Moving-boundary; Transition simulation

1. Introduction

Condenser is a major component in refrigeration system and air conditioning system, its performance has a direct effect on the system efficiency, such as the effect of service life and energy consumption. At present, due to its energy saving, frequency conversion air conditioning is gradually recognized by people.

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doi:10.1016/j.proeng.2012.01.1106

Its compressor isn't frequent start and stop, so the study should focus on short transient changes, namely the transient response of various boundary conditions, so as to provide optimization control strategy for frequency conversion air conditioning.

Here, a dynamic mathematical model of an air-cooling condenser based on the moving-boundary

Nomenclature

ρ	density (kg/m^3)	A	area (m^2)
h	enthalpy (kJ/kg)	L	length (m)
p	pressure (Pa)	Superscripts	
t	time (s)	–	mean value
α	convective heat transfer coefficient (W/(m^{2}k))	Subscripts	
T	temperature (℃)	1,2,3	regions of condenser
D	diameter of heat transfer tube (m)	i, o	inner, outer
C_w	wall specific heat (J /(kg K))	12,23	regions interface
f_o	outer area of condenser per meter of pipe length	in, out	inlet, outlet
η_o	fin tube efficiency	r	refrigerant
γ	void fraction of two-phase flow	a	air
m	mass flow rate (kg/s)	gb, lb	saturated gas, saturated liquid
u	velocity (m/s)	b	saturation
ΔT_1	degree of superheat (℃)	w	pipe wall

model has been developed, using the zone model in this paper. Moving-boundary model is frequently used to track the phase change of the refrigerants fluid in the heat exchanger, as to the condenser, it is the length change of the phase region, namely the superheated region, the two-phase region and the sub-cooled region. Moving-boundary method used in simulation of heat exchanger has been presented in many literatures[1-4], however, they rarely considered the time-variant void fraction, time-variant heat transfer coefficient together, moreover, they rarely discussed the heat exchange between the pipe and the air side[5]. Hence, the time-variant mean void fraction of the two-phase and the time-variant mean heat transfer coefficient of each region are taken into account simultaneously in the model in this paper, as well as the heat exchange between the pipe and the air side. In order to avoid the iterative calculation, all state parameters of the mathematical model are expressed as the functions of the temperature. The zone model constructed here will well predict the transient response of the condenser, when the compressor speed, air temperature and air quality flow are changed separately.

2. Condenser Model

An air-cooling condenser of finned tube is discussed in this paper. Due to the complexity of physical model and dynamic process, the primary assumptions for condenser are as follows:

- The refrigerant is one-dimensional flow and compressible fluid. The air is one-dimensional flow and incompressible fluid. The refrigerant and the air flow are regarded as a counter-current form[6].
- The pressure drop in the condenser and the axial heat conduction of pipe wall are neglected.
- The specific heat of air is constant. The energy storage in the air is neglected.
- There are three zones in the condenser at the very start.

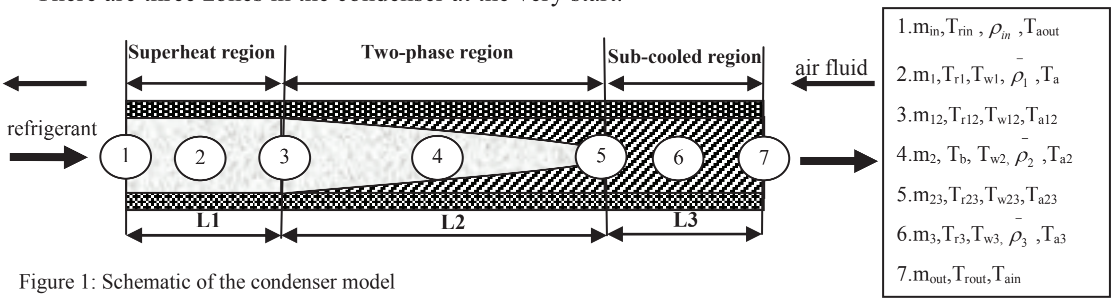

Figure 1: Schematic of the condenser model

There is the heat exchange between the refrigerant and pipe wall, as well as heat exchange between the air and pipe wall. According to above assumptions, the governing equations of the refrigerant flow for mass and energy can be expressed as follows[7]:

$$\frac{\partial \rho}{\partial t}+\frac{\partial(\rho u)}{\partial x}=0 \tag{1}$$

$$\frac{\partial \rho h}{\partial t}-\frac{\partial p}{\partial t}+\frac{\partial \rho u h}{\partial x}=\frac{4}{D_i}\alpha_i(T_w-T_r) \tag{2}$$

The energy equation for pipe wall and fins is:

$$C_w\rho_w A_w\frac{\partial T_w}{\partial t}=\alpha_i\pi D_i(T_r-T_w)+\alpha_o f_o\eta_o(T_a-T_w) \tag{3}$$

Given of the small mass and specific heat of the air, the mass and energy storage in the air can be neglected. Therefore, air side can use steady state equation which describes as below:

$$m_a\frac{dh_a}{dx}=\pi D_o\alpha_o(T_a-T_w) \tag{4}$$

when the refrigerant in the condenser is two-phase region ,the ρ and ρh in the above equations can express as follows:

$$\bar{\rho}=\bar{\gamma}\rho_{gb}+(1-\bar{\gamma})\rho_{lb}\ ,\quad \bar{\rho h}=\bar{\gamma}\rho_{gb}h_{gb}+(1-\bar{\gamma})\rho_{lb}h_{lb} \tag{5}$$

γ is the average void fraction in the two-phase region, it can reference the literature [1].

$$\bar{\gamma}=1-[1+(\rho_{gb}/\rho_{lb})^{2/3}(2/3\ln(\rho_{gb}/\rho_{lb})-1)]/[(\rho_{gb}/\rho_{lb})^{2/3}-1]^2 \tag{6}$$

3. Numerical Resolution

(1)- (4) are the basic equation of the dynamic model of the condenser. Simultaneous equations should be applied in each region of the condenser. Integration of the simultaneous equations over the superheated region, the two-phase region and the subcooled region are along the flow direction, from *0* to L_1, from L_1 to L_2, from L_2 to L_3, respectively. Leibniz's rule is employed on the simultaneous equations in each region. The Integral results of the superheated region give as below. And the computational detail of the other two regions is omited, due to its similar method to the superheated region.

$$L_1(\frac{\partial \bar{\rho}_1}{\partial T_b}-\frac{1}{2}\frac{\bar{\rho}_1}{\Delta T_1})\frac{dT_b}{dt}+\frac{1}{2}L_1\frac{\partial \bar{\rho}_1}{\partial \Delta T_1}\frac{dT_{rin}}{dt}+(\bar{\rho}_1-\rho_{gb})\frac{dL_1}{dt}+\frac{m_{12}-m_{in}}{A_i}=0 \qquad (7)$$

$$[L_1\bar{\rho}_1(\frac{\partial \bar{h}_1}{\partial T_b}-\frac{1}{2}\frac{\partial \bar{h}_1}{\partial \Delta T_1})+L_1\bar{h}_1(\frac{\partial \bar{\rho}_1}{\partial T_b}-\frac{1}{2}\frac{\partial \bar{\rho}_1}{\partial \Delta T_1})-L_1\frac{\partial p}{\partial T_b}]\frac{dT_b}{dt}+(\bar{\rho}_1\bar{h}_1-\rho_{gb}h_{gb})\frac{dL_1}{dt}$$

$$+\frac{h_{gb}}{A_i}m_{12}+\frac{1}{2}L_1\bar{\rho}_1\frac{\partial \bar{h}_1}{\partial \Delta T_1}\frac{dT_{in}}{dt}-\frac{m_{in}h_g}{A_i}-L_1\alpha_{i1}\frac{4}{D_i}(\bar{T}_{w1}-\bar{T}_{r1}) \qquad (8)$$

$$C_w\rho_w A_w\frac{d\bar{T}_{w1}}{dt}+(\bar{T}_{w1}-T_{w12})\frac{dL_1}{dt}=L_1\pi\alpha_{i1}D_i(\bar{T}_{r1}-\bar{T}_{w1})+L_1\pi\alpha_{o1}D_o(\bar{T}_{a1}-\bar{T}_{w1}) \qquad (9)$$

$$m_aC_a(T_{aout}-T_{a12})=\pi L_1\alpha_{o1}D_o(\bar{T}_{a1}-\bar{T}_{w1}) \qquad (10)$$

In the above equation $\Delta T_1=T_{rin}-T_b,\bar{\rho}_1=f(T_b,\Delta T_1)$, $\bar{h}_1=f(T_b,\Delta T_1)$, $\bar{T}_{r1}=(T_{rin}+T_b)/2$, $\bar{T}_{a1}=(T_{aout}+T_{a12})/2$. Integral calculation of the three regions contains 12 equations with 7 state variables: (L_1, L_2, T_b, $T_{rout}, \bar{T}_{w1}, \bar{T}_{w2}, \bar{T}_{w3}$). The dependent variable (T_{aout}) can be calculated from the state variables through the equations.

To solve the equations, initial conditions and boundary conditions must be added. Initial conditions of the dynamic model can be calculated by the steady-state simulation. Boundary conditions at the inlet of the condenser is set as the outlet of the compressor, and at the outlet of the condenser, the mass flow rests with the TEV. At the outside of the condenser coil, there is the air which acts as the cooling medium of the air-cooling condenser. Hence the boundary conditions on the heat exchanger surface depend on the air conditions which are artificial hypothesis in this paper. The boundary condition can be described as: $x=0,h_{in}=h_{cpo},m_{in}=m_{cpo}$; $x=L_1+L_2+L_3,m_{out}=m_v$; The m_a and T_{ain} are set to be some value.

4. Simulation and Discussions

The condenser is a twin circuit air-cooling coil with 52 tubes, the pipe length is 800 mm, inner diameter is 6.75 mm and the outer diameter is 7mm. The number of aluminum flat fin is 570.Rotor speed of the scroll type compressor is 2820rpm, and compressor cavity volume is 14.18cm^3. The coefficient of thermal expansion valves is 3.7835E-007. Based on the above structure parameter model, performance of fin and tube condenser with R22 was simulated to achieve a stable state first, and then changed the boundary conditions. These transient conditions include the compressor speed, the air inlet temperature and the air inlet flow rate. When the simulation was doing, one of the above conditions was changed and the other two conditions remain unchanged. At t=9 s the compressor speed ω was increased by 63, at

Table 1: Steady-state results

steady-state results				
L_1=2.62	L_2=17.37	L_3=2.71	C_w =385	m_{ain} =0.233
ρ_w =8960	T_{ain} =34	T_{r1} =80	T_{r2} =51	T_{r3} =45
α_{i1} =623	α_{i2} =1455	α_{i3} =426	α_{o1} =59	α_{o2} =60
α_{o3} =60	P_b =1.9965E+6	$\omega = 60$	Q=2.8824	m_{in} =0.0129

t=200s the air inlet temperature was increased by 35.7℃ and at t=400s the air inlet flow rate was increased by 0.244 kg/s. The transient response of condenser pressure and heat exchange quantity isshown in Fig.2 and Fig.3 , respectively. The Fig.2 shows that other parameters are constants : (a) When the compressor speed is increased, the refrigerant storage in the tube increases, so that the pressure of the condenser rises. (b) When the air inlet temperature is increased, the pressure of the condenser rises due to the reduction of heat exchange.(c) When the air inlet flow rate is increased, the pressure of the condenser reduces for the enhancement of heat transfer. The Fig.3 shows that other parameters are constants : (a) An

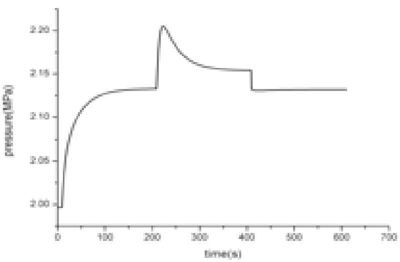

Figure 2: Pressure in the condenser

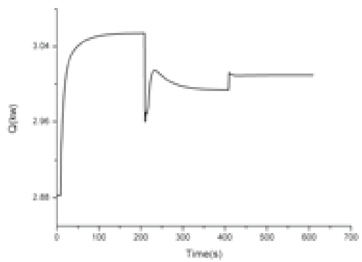

Figure 3: Heat exchange in the condenser

increase of compressor speed as well as an increase of air inlet flow rate result in more heat exchange.(b)An increase of air inlet temperature results in less heat exchange. These simulation results obtain the right trends comparing with other related literature.

5. Conclusions

A zone transient model for predicting air-cooling condenser performance has been developed in this paper. New features of the model include the heat transfer between the air and fin , the air and tube are took into accounted and considered into the steady state, state parameters are all expressed as the functions of the temperature .The response of the condenser to varying the compressor speed, the air inlet temperature and the air inlet flow rate is studied and yields expected trends using this model. The simulation speed and rationality of the model will provide certain reference value for the optimization design of the control strategy in the refrigeration and air conditioning system.

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